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HEAT RECOVERY IN LIVESTOCK AND POULTRY BUILDINGS

Six important questions

ENERGY TECHNOLOGY SERIES

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ENERGY TECHNOLOGY
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PUBLICATION 88-4000 available from
Agricultural Energy Centre
Guelph Agriculture Centre
P.O. Box 1030
Guelph, Ontario
N1H 6N1
Publication Date May 1988

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INTRODUCTION

This booklet provides a fundamental overview of heat recovery, and is intended primarily for livestock producers who wish to determine if heat recovery might benefit their animal facilities.

The booklet incorporates principles of ventilation heat recovery so as to provide farm operators with a basic, straightforward understanding of this science.

Many operators are not conversant with engineering procedures such as psychometrics and heat/mass transfer calculations, and it is not the purpose of this booklet to make you an expert. Consequently, such subjects have been deliberately omitted. However, since these subjects are essential in the actual analysis of heat recovery systems, general recommendations and a computer program — both of which embody the required engineering principles — have been substituted. The recommendations are included in the booklet; the computer program is readily available to any livestock producer. When you combine these with the answers to the six questions that appear in the booklet you will then be ready to make an initial appraisal of ventilation heat recovery and its suitability for your barn.

1. WHAT IS HEAT RECOVERY?

1.1 The Concept

Heat recovery is, as the name suggests, the capturing and subsequent utilizing of heat that would otherwise be rejected, (i.e. wasted). In agriculture there are a number of ways in which we can recover valuable heat. For example, the heat that is removed from milk by the refrigeration system in a dairy operation can be recovered and used to warm wash water.

Ventilation air exhausted from livestock buildings is anther source from which heat may be recovered. In Ontario, the reason for doing so is simple: during the winter our barns frequently require extra heat.

Heat is recovered from exhaust air by transferring it, often across a thin solid sheet of material, to the incoming (fresh) air that replaces the exhaust air being discharged from the barn. There are various designs, but they all have the same basic function; namely, the transfer of heat from a warm airstream to a colder one. In some cases the heat is transferred to an intermediate fluid such as glycol before being delivered to the barn air. Figure 1-1 shows how the heat is transferred in a plate type exchanger.

Note carefully that only heat is transferred across the heat exchange surface. None of the dust, noxious gases or moisture contained in the exhaust air get into the fresh incoming air with true heat recovery. When contaminated exhaust is re-introduced into the barn by mixing with fresh supply air the process is referred to as *recirculation*.

1.2 Types of ventilation air heat recovery systems

There are many designs and system configurations for recovering heat from warm exhaust air, including the following:

- plate type exchangers (cross-flow, parallel-flow and counter-flow);
- shell and tube exchangers (cross-flow, parallel-flow and counter-flow);
- regenerative exchangers, i.e. recycling or reversing cycle types;
- thermosiphon exchangers;
- open runaround (liquid spray) systems; and
- closed loop runaround systems.

At present, most existing agricultural heat recovery installations are of the plate or shell and tube design, with plate type exchangers predominating. Farm operators may choose to either purchase a manufactured exchanger, or construct one themselves.

With the exception of open runaround and heat wheel systems (both of which are sometimes used for industrial applications) all of the other heat exchanger types have been tested successfully in livestock facilities. However, since few, if any, thermosiphon or closed loop exchangers are commercially available (i.e. in pre-manufactured form) and since these do not readily lend themselves to on-farm fabrication, their use in agriculture is limited. The majority of manufactured exchangers and most of the available plans for self-built exchangers are based on the plate type design, although there is considerable variation in how the plate principle is employed. If you are interested in evaluating or possibly using other types of exchangers you will find the additional literature list helpful. As well, a qualified agricultural engineer can provide further information concerning such equipment.

Plate type air to air heat exchangers

Plate type exchangers involve the transfer of heat across either a single thin, solid plate (septum), or a series of plates. A single pane of glass can be used to illustrate how a plate exchanger works. During cold weather, heat is conducted through the glass, from your warm home and into the cold outdoors. No air is transmitted through the glass; only heat.

If you were to place a fan by the window in such a way as to blow a stream of air across the pane of glass, and if the wind were blowing briskly across the other side, then the rate at which heat would "cross" the pane would be much greater than would be the case if the fan were shut off and calm outdoor conditions prevailed. This principle is employed in most heat exchangers. A fan is used to propel exhaust air across one side of the exchange surface; another fan pushes cold incoming air across the other side. In so doing, heat is transferred across the plate — exactly as it is through your window.

As mentioned, there are two different methods or approaches for plate-type heat recovery in agricultural buildings. The first involves a single plate, such as thin plastic which, in order to provide sufficient heat transfer area must be fairly large (in the order of 12 m x 0.6 m, or 40 ft. x 2 ft.), as shown in Figures 1-2, 1-3 and 1-4. Under test conditions, in a chicken broiler barn, an exchanger based on this design recovered about 2 kilowatts (6,800 BTU/h) of heat during sub-zero weather conditions.

An alternative approach to using a single large heat transfer "plate" to obtain enough heat transfer area is to employ many smaller plates, "sandwiched" together – thereby creating a "matrix" of air passages – as shown in Figure 1-5.

In most plate exchangers of this design, exhaust air is forced through every second plenum — or passageway — while fresh air is conveyed within the other passages. Figure 1-1 shows a close-up view of three passages; one exhaust (in the centre) and two fresh air plenums (one on each side of the exhaust passage).

In Figure 1-5, the top fan forces stale barn air into the exhaust passages, the entrances to which are located in the top compartment of the exchanger. Fresh air is pulled through the corresponding supply passages by the bottom fan. This operating mode is shown diagrammatically in Figure 1-6.

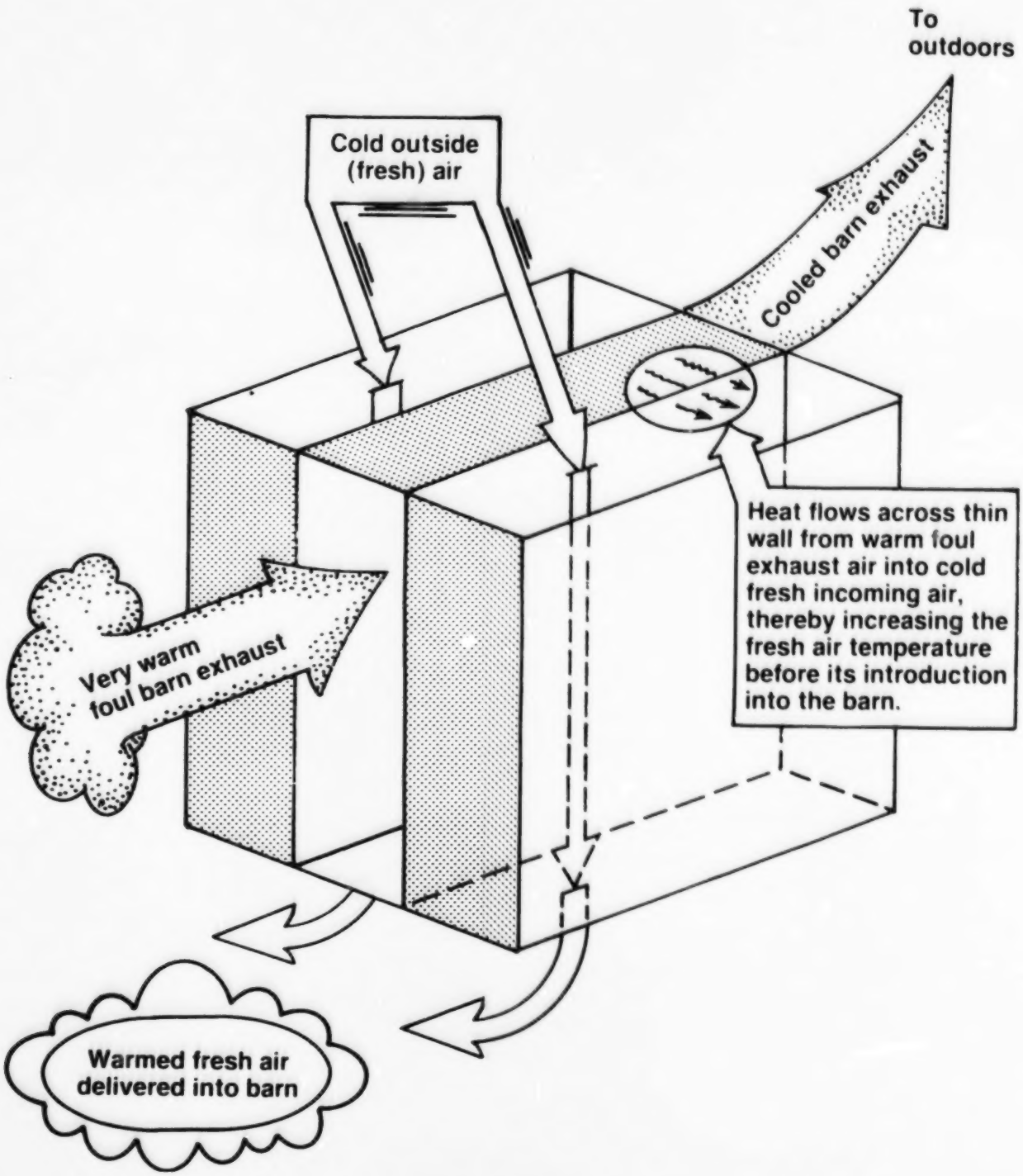

To outdoors
Cold outside (fresh) air
Cooled barn exhaust
Heat flows across thin wall from warm foul exhaust air into cold fresh incoming air, thereby increasing the fresh air temperature before its introduction into the barn.
Very warm foul barn exhaust
Warmed fresh air delivered into barn

Figure 1-1: *Cut-away section of a plate type air-to-air heat exchanger, showing the transfer of heat from the warm to cold air streams.*

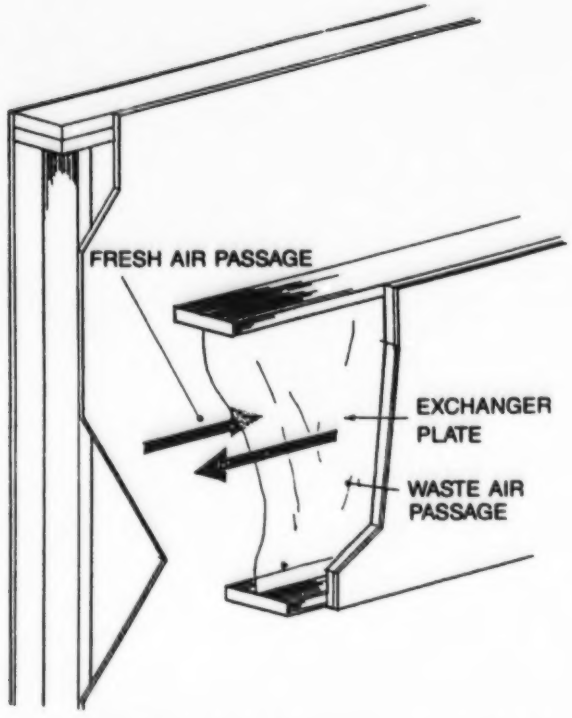
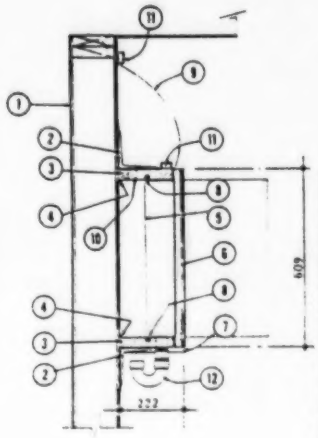


Figure 1-2: *Single plate heat exchanger section. (ref. 1)*



1.	EXTERIOR WALL.
2.	150x150 mm SHELF BRACKETS.
3.	38x184 mm TOP & BOTTOM FRAME OF EXCHANGER WITH 12.5x12.5 mm CONTINUOUS GROOVE.
4.	CAULKING TO PREVENT AIR LEAKAGE.
5.	150 µm POLYETHYLENE HEAT EXCHANGER PLATE.
6.	INTERIOR EXCHANGER COVER 25.4x609 mm POLYSTYRENE INSULATION, GLUED TO 12.5x609 mm PLYWOOD.
7.	19x38x222 mm SUPPORTS, SECURED TO 38x184 mm BOTTOM FRAME @ 600 mm o.c., TO SUPPORT EXCHANGER COVER DURING INSTALLATION.
8.	12.5x12.5 mm FILLER TO SECURE TO 38x184 mm TOP & BOTTOM. USE STAINLESS STEEL SCREWS.
9.	AIR DISTRIBUTION DUCT.
10.	FRESH AIR PASSAGE OUTLETS TO AIR DISTRIBUTION DUCT.
11.	19x38 mm CONTINUOUS TO SECURE 150 µm POLYETHYLENE.
12.	38 mm SINK FLANGE & P-TRAP DRAIN IN WASTE AIR PASSAGE IN TWO LOCATIONS OF EXCHANGER.

Figure 1-3: *Heat exchanger construction. (ref. 1)*

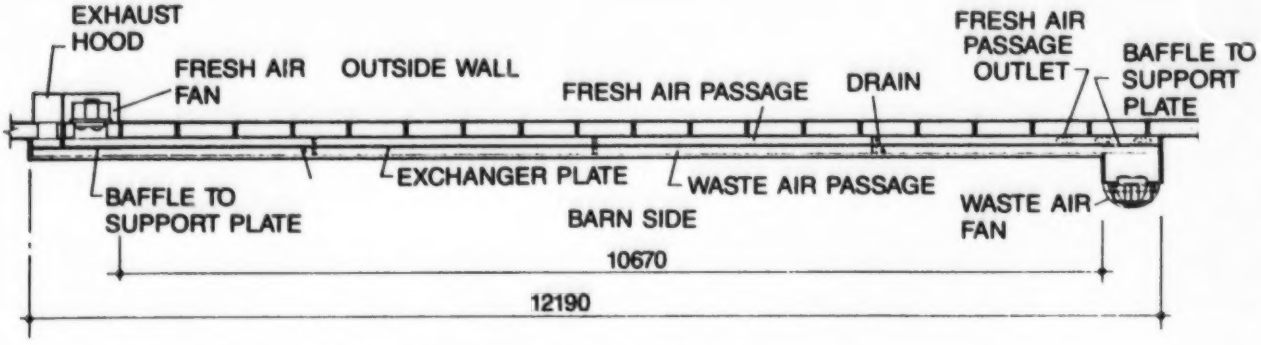


Figure 1-4: *Plan view of single plate heat exchanger. (ref. 1)*

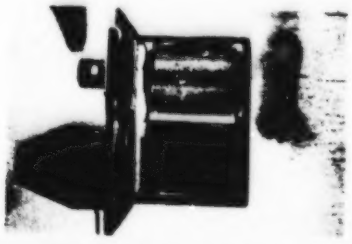


Figure 1-5: *Front view of multiple plate heat exchanger. Exchanger front face is hinged to facilitate access to core for cleaning. (ref. 5)*

The entire front portion of the exchanger, containing both fans, is hinged and swings open to permit cleaning.

The core — composed of many individual plates — is often constructed of plastic, although metal cores (e.g. aluminum) are also used.

The particular exchanger shown in Figure 1-5 was, under test conditions, able to recover 3.8 kW (12,900 BTU/h) from 18°C (65°F) exhaust air when the outside air

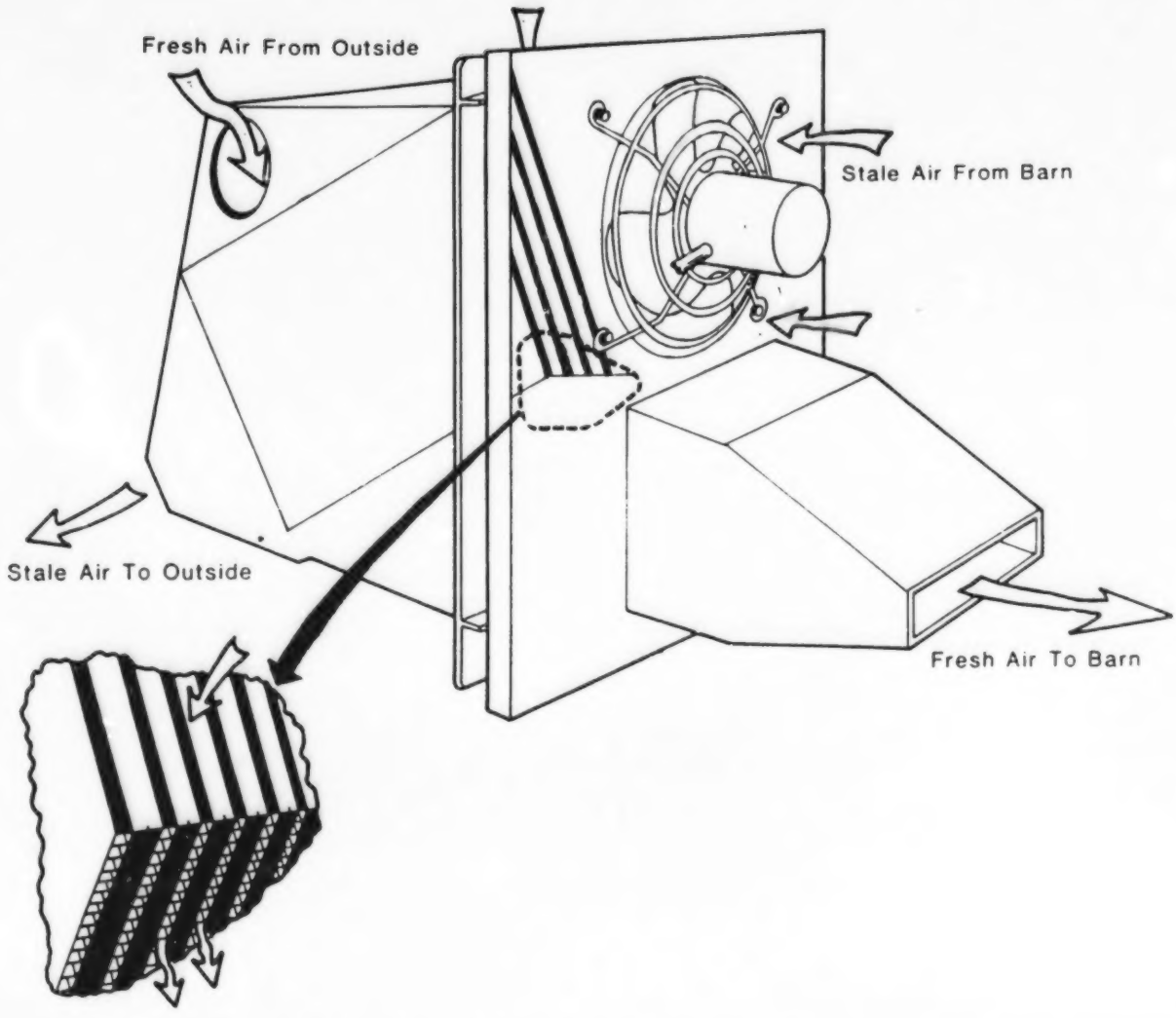


Figure 1-6: *Air flow patterns in typical cross-flow heat exchanger. (ref. 5)*

temperature was −15°C (+5°F). Consequently, fresh air entered the barn at 8°C (46°F) as opposed to −15°C (+5°F).

Shell and tube heat exchangers

As in the case of plate exchangers, shell and tube type heat exchangers warm incoming fresh air by transferring the heat contained in exhaust air across a thin membrane or wall, but in this case, instead of a flat plate, a round tube or pipe serves as the heat transfer element — as shown in Figure 1-7. The exhaust air may flow within the pipes, with fresh air then being conveyed around their outside, or vice versa. Since barn exhaust air is often very dusty, the best approach is to convey it through whichever side of the tubes is most easily cleaned.

As in the case of plate exchangers, there are two ways to obtain sufficient surface area for the shell and tube design: either use one large area transfer surface or many

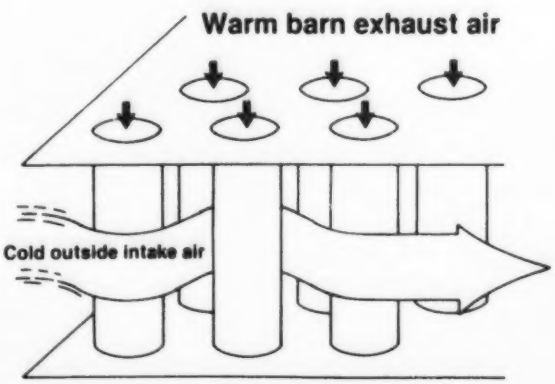


Figure 1-7: *Shell and tube heat exchanger, exhaust air flowing through tube side, fresh air across "shell" side.*

smaller surfaces. An example of the latter may be seen in Figure 1-8. This design employed 196 - 50 mm (approx. 2 inch) tubes, each of which was about 1,500 mm (5 feet) in length. The device recovered roughly 6 to 9 kW (20,000 to 30,000 BTU/h) during 0°C (32°F) outdoor and 8°C (46°F) indoor conditions.

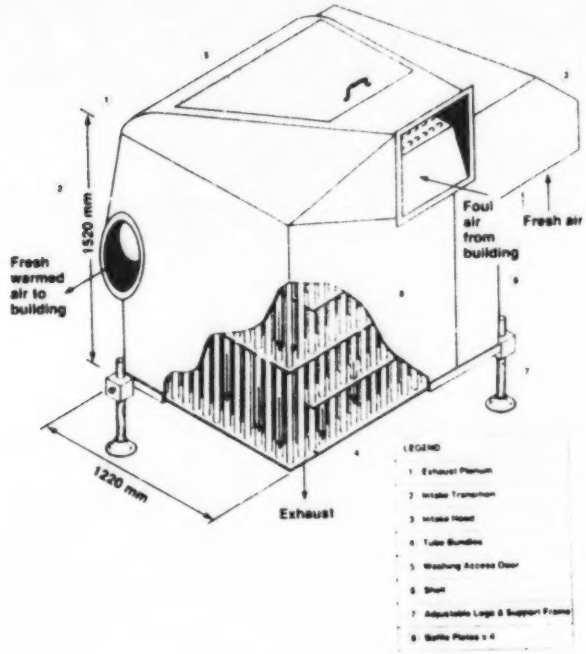


Figure 1-8: *Prototype 2360L/S (1100 cfm) shell and tube heat exchanger for confined livestock housing ventilation systems. (ref. 2)*

Figures 1-9 and 1-10 illustrate the alternative shell and tube design. In this case four large transfer surfaces, in the form of plastic air bags are used — as opposed to many smaller pipes. This arrangement has numerous advantages, including:

(i) contaminated air may be exhausted from (and fresh air delivered to) any number of locations rather than a single one as is the case with most other "single-point" designs.

(ii) there are no small, i.e. less than 15 mm (0.6 in.) passages that may clog with dust.

(iii) existing exhaust fans may be used to deliver stale exhaust air into the heat recovery device, thereby lowering capital costs.

The main drawback is the capital cost of constructing the additional building space within which the heat is recovered.

Figures 1-11 to 1-13 show a sideroom heat recovery system installed in a swine facility. This particular arrangement involves four fresh air supply tubes, extending along the entire 46 m (150 foot) length of one side of the barn. Stale air from all sow farrowing and weaner rooms is first exhausted into the sideroom recovery chamber, whereupon a portion of its heat is transferred across the tube walls to the fresh air. The cooled exhaust air then leaves the side room recovery chamber, being displaced by the steady supply of barn exhaust air. The warmed incoming fresh air is delivered to a "preheat" hallway from which individual rooms are supplied as required by their exhaust fan(s).

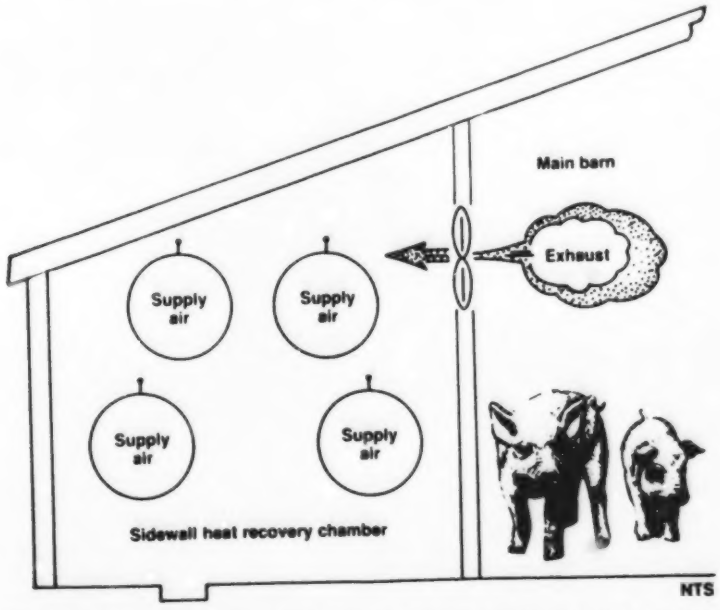


Figure 1-9: *Cross-section of typical side room heat recovery system, employing 4-61 cm (24 inch) suspended air bags for supply. Side room width is typically 2.4 m (8 feet) and side room extends along length of building.*

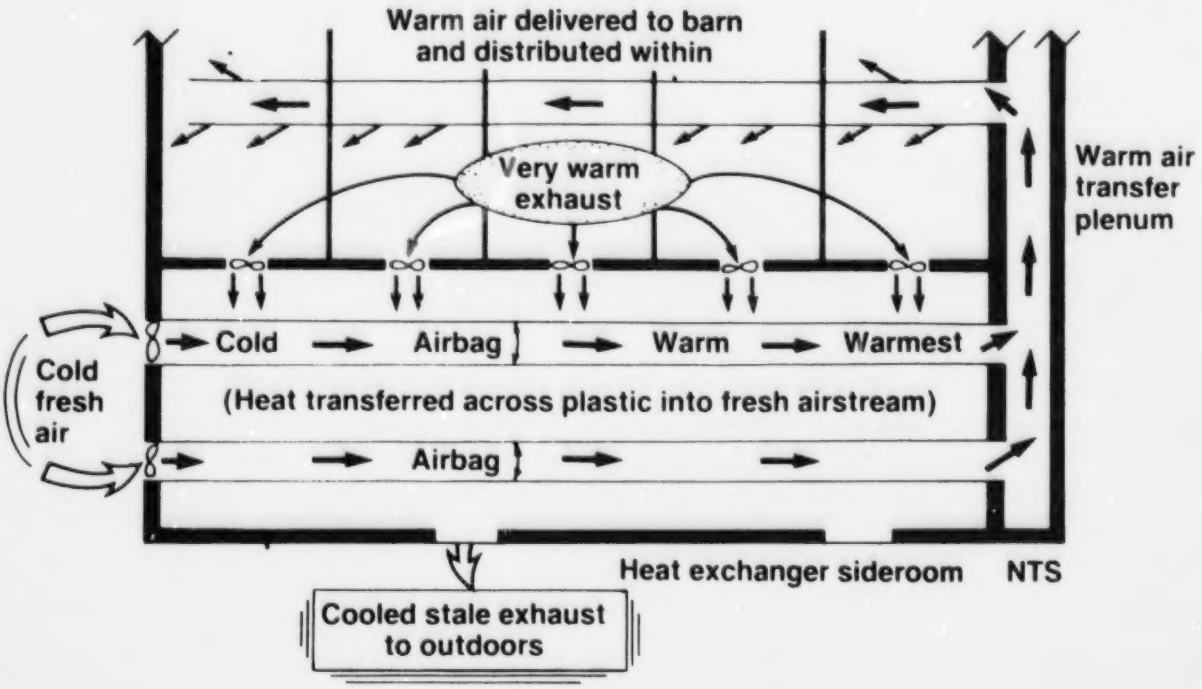


Figure 1-10: *Schematic diagram (plan view) of side room heat recovery system, using air bags to deliver fresh air.*

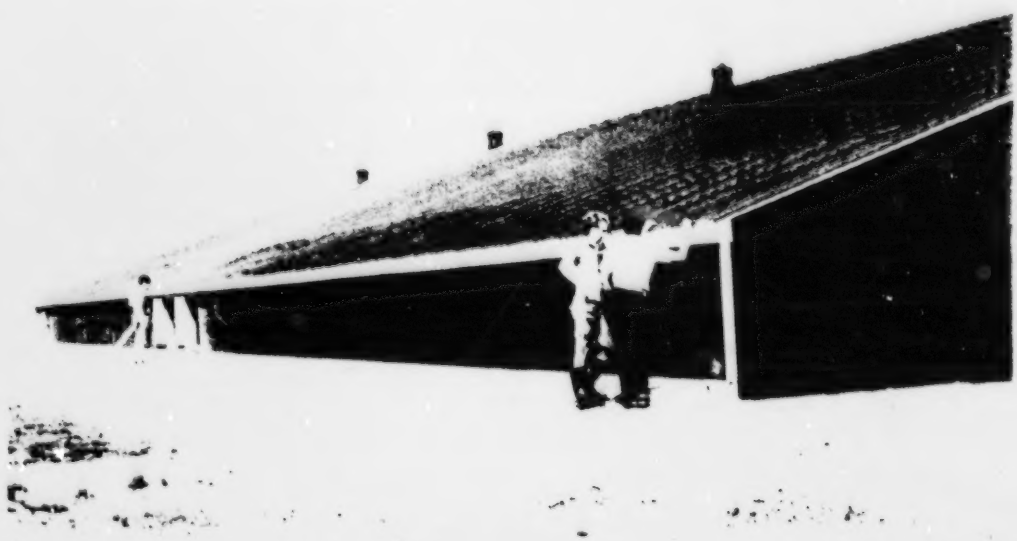


Figure 1-11: *Exterior view of side room heat recovery chamber on swine barn.*

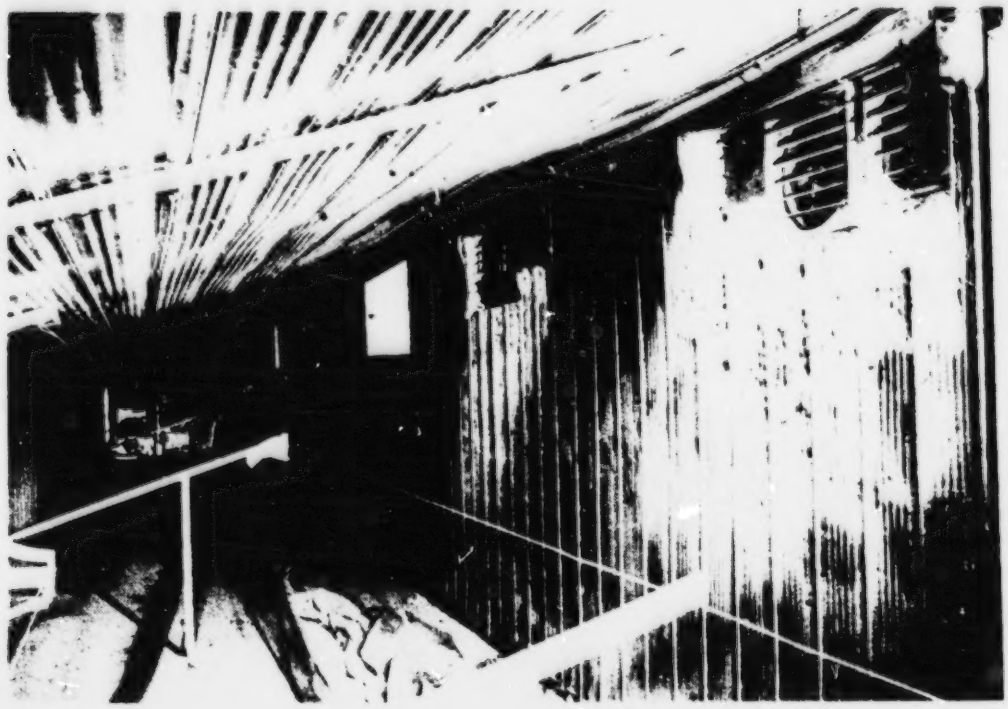


Figure 1-12: *Interior view of same side room heat recovery chamber — looking toward inlet end of air tubes. Plastic tubes are deflated and may be seen on floor. The system remains in this condition during summer when heat recovery is not used.*

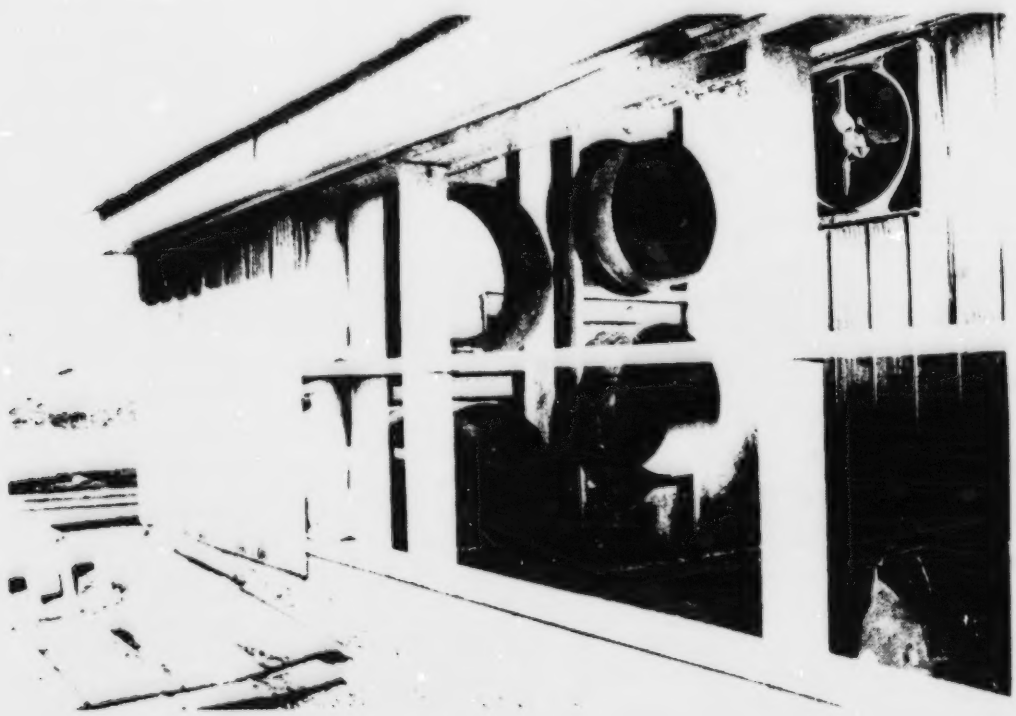


Figure 1-13: *Close-up view of air inlet end of side room heat exchanger. Note the four metal collars to which the plastic air tubes are connected during winter operating period.*

During testing, the temperature of the fresh air flowing inside the tubes was increased from 1.5°C (35°F) to 7.5°C (46°F), thus providing about 17 kW (58,000 BTU/h) of additional heat to the barn.

The majority of condensation was observed to occur along the initial portion of the fresh air tube. Consequently, the engineer involved with this particular design recommended that galvanized pipe be used, instead of plastic, for the first 4 m (13 ft.) of the tube, so as to avert possible damage arising from ice formations on the thin plastic.

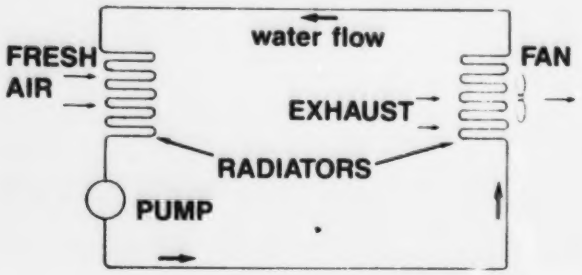


Figure 1-14: *Schematic diagram of closed loop runaround exchanger. (adapted from ref. 4)*

Closed loop runaround system

Closed loop runaround systems are at present, found infrequently in agriculture. Testing has been comparatively limited although the results to date are positive. Figure 1-14 depicts the manner in which these systems work. Heat is extracted from the exhaust air stream and transferred to a liquid by means of a simple heat recovery core, resembling (or sometimes, in fact, being) an automobile radiator. The warm liquid is thereafter pumped to another coil, over which cold fresh air is directed. The warm liquid releases its heat to the air in the same way as occurs in an automobile. Figure 1-15 shows the exhaust portion of a runaround system.

Past experience has shown that heat exchange coils equipped with fins are extremely prone to fouling, especially in poultry and swine facilities. Special attention must be devoted to the design of the coil, as adequate fin separation is critical.

Note that this system is capable of transferring the heat (i.e., the warmed liquid) a considerable distance from the exhaust location, unlike most air to air single-point exchangers.

Regenerative exchangers

Regenerative exchangers differ significantly from most others in that, rather than transferring heat across a wall to another fluid or gas, they store heat in a thermal mass and subsequently retrieve and deliver it to the barn by reversing the flow of air. This is illustrated in Figure 1-16. This unit reverses the fresh and exhaust air flow directions at 10 minute intervals. Thus the cold air is constantly being warmed by the heat imparted to the bed of rocks by the exhaust air during the preceding cycle.

This exchanger was operated successfully in a finishing barn, located in Saskatchewan, and provided all of the energy that would otherwise have been supplied as supplementary heat.

Exchangers such as these will, in all likelihood, be constructed on-farm. Certain potential drawbacks exist with these units, as follows:

(i) Humidity, microorganisms and dust may possibly be returned to the barn, having been first deposited on the rock bed during the exhaust portion of cycle, and then picked up by the fresh air during the supply phase.

ii) An improper ratio of supply to exhaust air flows will cause not only condensation of water on the cold rocks, but also freezing. This in turn will block all the inter-rock air spaces and effectively stop the exhaust air flow.

(iii) Excessive fouling of the rock bed with dust and/or feathers might also restrict air flow, so provision for cleaning should be made. Even if filters are employed, fine dust can cause problems (see section 4.4).

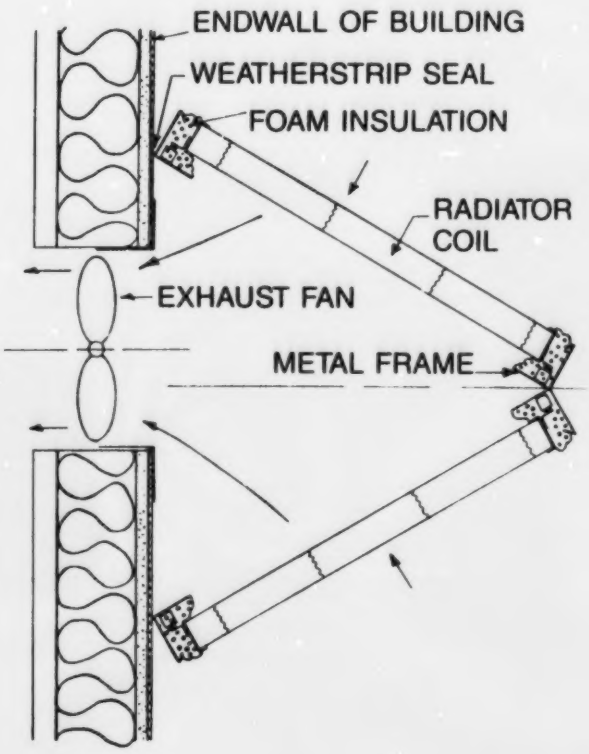


Figure 1-15: *Horizontal section of exhaust unit as installed in swine nursery. (adapted from ref. 4)*

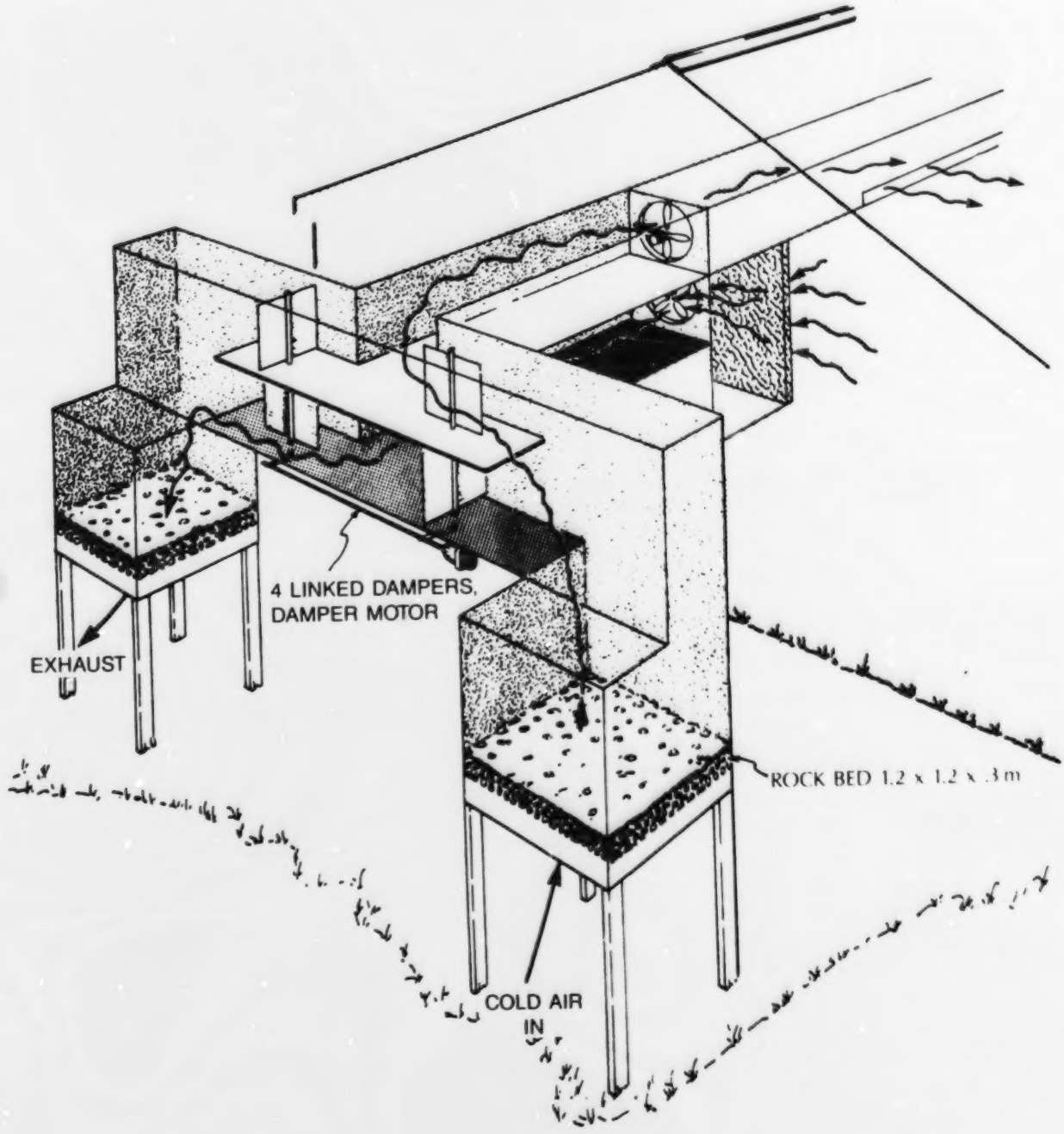


Figure 1-16: *Rock bed regenerative heat exchanger. Air flow directions are periodically reversed by changing the damper position. Rock size is approximately 7.5 cm (3 inches). (ref. 8)*

2. WHY MUST WE KEEP LIVESTOCK COMFORTABLE? (BALANCING THE BARN)

2.1 The reasons — performance and profits

Most livestock types can tolerate a wide range of environmental conditions (temperature, humidity, noxious gas concentrations etc.). However, there is a comparatively narrower range in which they will perform optimally, and optimum performance almost invariably means maximum profits. In simple terms, if the animals thrive so do the producers.

Outside weather conditions don't always make it easy to maintain near-perfect conditions within the barn and so we turn to technology and improved management practices to achieve this. Heat recovery is one of many ways by which barn conditions can be improved — at potentially lower cost.

2.2 Thermoneutral range

Every livestock type has a specific comfort zone — or temperature range within which they are neither shivering due to cold nor subject to the dehydrating effects of overheating. This zone is called the **thermoneutral range** within which animals will perform best.

If the barn temperature *exceeds* the thermoneutral range the animals will go off their feed, consume large quantities of water and ultimately perish in extreme heat.

Conversely, if the temperature is *below* the thermoneutral range they shiver and often crowd together. Also, in this case, feed consumption will increase and feed conversion (i.e. the quantity of feed consumed per kg of meat, or per dozen eggs) will suffer. In otherwords, feed efficiency *drops* as the animals eat more feed energy which they in turn convert to heat energy, rather than food energy. Their bodies actually produce — and then give off — more heat as they try to hold their internal body temperature at the normal point. In extreme cold, the animals will be unable to do so and the internal body temperature will drop, causing a condition called hypothermia.

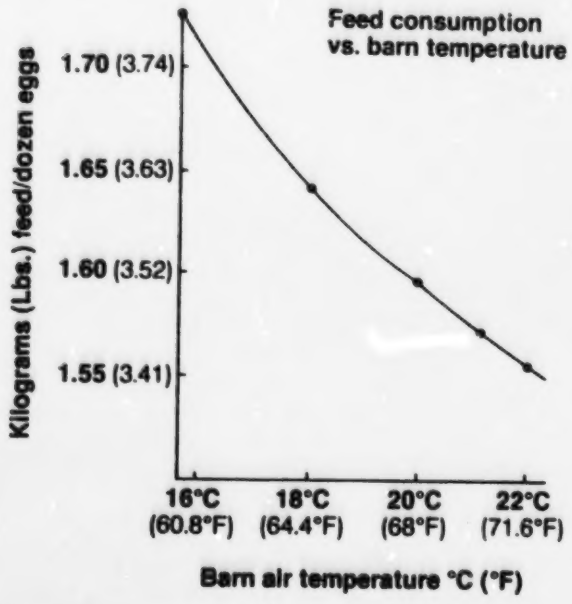


Figure 2-1: *The effect of temperature on feed consumption for layer hens.*

If the air temperature drops slightly below the thermoneutral point, mortality is unlikely but livestock will consume more feed, thus cutting into profits. Secondly, health problems may ensue as a result of low temperature stress. Symptoms include coughing, digestive upsets, increased disease — especially respiratory ailments — and generally, unthrifty appearance and performance, thereby causing a further reduction in profits.

Researchers have actually measured how much extra feed is consumed, and how greatly conversion is affected by sub-optimum barn temperatures. This is shown for swine in Tables 2-1 and 2-2, and is illustrated graphically for layer hens in Figure 2-1.

TABLE 2-1. EFFECT OF AIR TEMPERATURE ON RATE OF GAIN WITH SWINE (REF. 8)

Mean live weight, kg	Average daily gain, kg/(pig-day), at air temperatures of						
	5°C	10°C	15°C	20°C	25°C	30°C	35°C
45		0.63	0.71	0.87	0.90	0.73	0.40
68	0.58	0.67	0.79	0.95	0.87	0.64	0.22
91	0.55	0.72	0.85	0.99	0.84	0.55	0.03
113	0.52	0.76	0.92	0.96	0.78	0.45	-0.15
136	0.50	0.80	1.00	0.95	0.72	0.35	-0.36
159	0.47	0.86	1.05	0.93	0.67	0.26	-0.55

TABLE 2-2. EFFECT OF AIR TEMPERATURE ON FEED CONVERSION WITH SWINE (REF. 8)

Live weight, kg	Feed conversion, kg feed/kg grain, at air temperatures of						
	5°C	10°C	15°C	20°C	25°C	30°C	35°C
32 to 65	4.8	4.4	3.7	2.8	2.6	5.5	7.8
75 to 118	10.0	5.1	3.7	4.0	4.2	9.0	

A helpful (and recommended) exercise is to calculate the present feed conversion in a given livestock production facility and then compute how much feed could be saved if air temperatures were maintained at the optimum level.

2.3 Thermal (energy) balance — the tightrope

To understand why heat recovery may be useful in your barn you should first know how the barn is "balanced" in terms of energy. To illustrate this, let's refer to the energy balance equation for a barn, which is as follows:

$$Q_A + Q_S - Q_C + Q_V$$

(Animal heat) + (Supplementary heat) − (Conducted heat) + (Ventilated heat)

Heat gains − Heat losses

$$\Delta$$

The four heat components in the above equation are as described below:

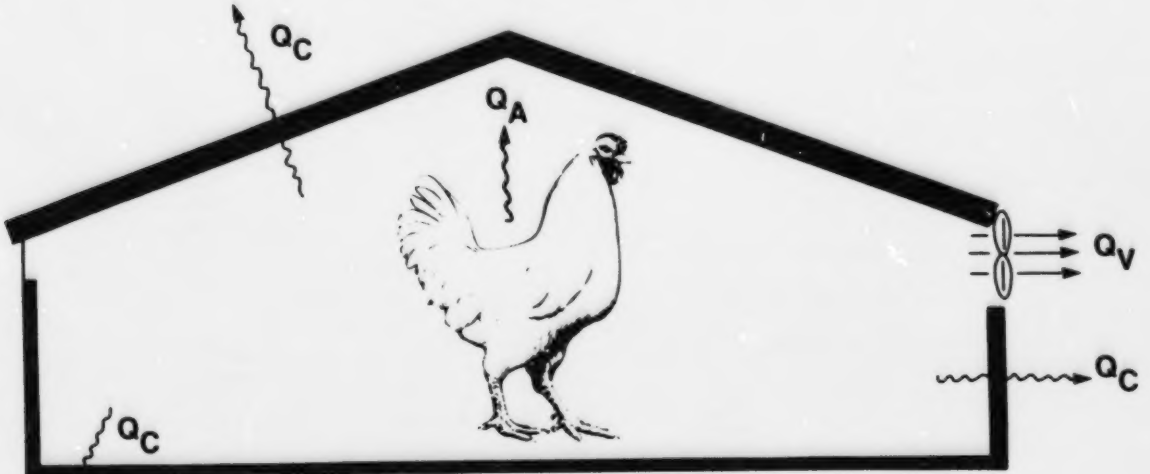
- Q_A – *Heat from the animals* (derived originally from feed)
- Q_S – *Supplementary heat* required to maintain barn air temperature in the thermoneutral range. It is usually derived from either electricity, wood fuel, propane, oil, natural gas, but it may also be **recovered** from exhaust air.
- Q_C – *Conduction heat loss* which is the heat that passes directly through the solid building structure, i.e. through the ceiling, walls, foundation and floor. Heat flows from hot to cold areas, and the colder it gets outside, the faster heat will flow through the walls. If insufficient heat is available in the barn (from livestock and heaters) to make up for the higher loss rate then the air temperature will fall, or the ventilation rate will drop — until a new equilibrium or balance point is reached.
- Q_V – *Ventilation heat loss.* Exhausted barn air contains dust, moisture (from manure and animal respiration), harmful gases such as H_2S, CO_2 ammonia, and, by necessity, heat. In fact, in the order of 75% of the heat lost from most barns during the winter is via the ventilation system, while only 20 to 30% is conducted through an insulated building shell. Since what goes out must come in, an amount of fresh cold air — equal to the exhaust — must be introduced into the barn to displace or "make up" for the exhaust air. Like Q_t, Q_v becomes larger as the difference between barn and outside air temperature increases (assuming the same number of fans are kept running). Note that switching on more fans will also increase Q_v since more cold air will be coming into the barn.

The energy equation invariably balances, but not always at the barn temperature we would like, as depicted in Figure 2-2.

For example, if it becomes very cold outside, Q_t and Q_v go up. If the quantity of animal heat is insufficient to make up for barn heat losses, then the barn temperature drops. Q_t and Q_v then decline because, as mentioned earlier, they are proportional to the *difference* between the barn air and outside air temperatures.

An extreme example of a potential heat imbalance can be seen with newborn animals such as piglets, poults or chicks. Their heat output (Q_A) is very small, and failure to maintain adequate temperature causes mortality. Supplementary heat in this instance is therefore essential.

Some growers endeavour to maintain correct barn temperature by lowering the ventilation rate (e.g. by operating fewer fans), but this usually causes problems — as we will see in the next section.



Outside temperature = –5°C (23°F)
Desired barn air temperature = 22°C (72°F)

Q_A (animal heat)	67.5 kw (230000 Btu/h)
$-Q_C$ (conducted heat loss)	20.5 kw (70000 Btu/h)
$-Q_V$ (ventilation heat loss)	63.0 kw (215000 Btu/h)
Heat Balance (Deficit)	–16.0 kw (55000 Btu/h)

∴ Barn air temperature drops to approximately 17°C (63°F)
Result: feed consumption increases, conversion efficiency decreases. (adapted from ref. 6)

Figure 2-2: ***Energy budget (without heat recovery)***

A preferable way to balance the equation at the desired temperature is to give Q_A and Q_S some additional "help", so they may match the magnitude of Q_C and Q_V.

3. HOW DOES VENTILATION PRODUCE PROFITS AND HOW MUCH VENTILATION IS NEEDED?

3.1 Benefits

As explained in section 2.3, adequate ventilation removes moisture (humidity) and harmful by-product gases and contaminants from the barn environment. This also preserves the building, by keeping the inside air "dry" enough to minimize condensation — or sweating — on the barn's inner surface, thus preventing excessive rusting and rotting of building materials. Ventilation also keeps the herd or flock (and the farm operator) healthy by maintaining gases, and dust at a sufficiently low concentration in the barn air. As the air is exhausted, fresh air, containing oxygen (O_2) is brought into the barn.

As well, during warmer weather, the ventilation system discharges surplus heat from the barn, thereby maintaining the temperature balance shown in the balance equation.

Since heat recovery is only needed during cool or cold weather we will focus solely on winter ventilation rates.

3.2 Winter Ventilation

The recommended amount of ventilation (usually expressed in litres per second, L/s, or cubic feet per minute, c.f.m.) is dependent upon the number, size and type of livestock, as well as outdoor temperatures. In some cases, the amount of ventilation needed for moisture control will differ from that needed to ensure that harmful gas concentration levels are held below the point at which they will impair herd or flock health.

Table 3-1 gives recommended minimum winter ventilation rates, but note that these are often "starting points" for the operation of many ventilation systems. For example, in the case of a poorly sealed barn, uncontrolled air leakage may necessitate a lower mechanical ventilation rate. Conversely, in other barns (which are not necessarily tightly sealed) some operators have found it necessary to augment the ventilation rate — beyond the minimum — because the air feels "stuffy", "close" or "stale". In most cases there is a valid reason for this action. Often what is occurring is that the noxious gas levels are too high, or additional moisture is being released from sources other than the animals themselves, e.g. a manure pack, droppings pit.

A further "rule of thumb" often employed is to provide a minimum ventilation rate of **four** air changes per hour (4 ACH). For example, the barn used for the calculations

TABLE 3-1: MINIMUM RECOMMENDED VENTILATION RATES FOR WINTER VENTILATION

Type of Livestock	Housing System	Minimum Ventilation Rate*
Dairy Cows	Free Stall	19 L/s per 450 kg**
	Stanchion (well insulated)	24 L/s per 450 kg
Calves		
50 kg (average)	Continuous	6.6 L/s per calf
63 kg (average)	Continuous	9.5 L/s per calf
45 kg (at start)	Batch	4.7 L/s per calf
140 kg (at finish)	Batch	11.3 L/s per calf
Beef	Free Stall	19 L/s per 450 kg
Poultry - Hens	Confined (well insulated)	.5 L/s per bird
Pullets	Confined (well insulated)	.5 L/s per bird
1.8 kg Broilers	Confined (well insulated)	.5 L/s per bird
Turkeys	Confined (well insulated)	16.5 l/s per m^2
Swine - Dry Sows	Confined (well insulated)	9.5 L/s per sow
Farrowing Room	Confined (well insulated)	14 L/s per sow+litter
Weaners	Confined (well insulated)	1.9 L/s per pig
Growers	Confined (well insulated)	2.4 L/s per pig
Finishers	Confined (well insulated)	2.8 L/s per pig
Growers-Finishers	Confined (well insulated)	2.8 L/s per pig
Horses	Free Stall	19 L/s per 450 kg
Sheep	Free Stall	1.9 L/s per 45 kg

1.0 L/s (litres/second) = 2.1 cfm (cubic feet/minute)
**1.0 kg = 2.2 lb.*
Source: Canadian Farm Building Code 1977

in figure 2-2 has dimensions of 76.2 m x 9.8 m x 2.4 m (250 ft. x 32 ft. x 8 ft.). Its volume is then 1,792 m^3, i.e. 1,792,000 litres (about 64,000 cubic ft.), and so 7,169 m^3 (256,000 cu. ft.) of air must be exhausted hourly. The corresponding ventilation rate is then 1,990 L/s (roughly 4,220 c.f.m.).

If poor air quality (e.g. an excessively high ammonia gas concentration level) periodically plagues your barn, determine whether your ventilation system is providing 4 ACH. If not, you may wish to consider providing this rate in order to keep gas concentrations (especially CO_2) at a safe and healthy level. CO_2 concentration, for example, should be no more than 0.26% (by weight).

3.3 HDT: The winter Heat Deficit Temperature

For every barn, except brooding units, there is an **outdoor** temperature, at which animal heat (Q_A) produced in the barn will exactly equal the conduction and ventilation heat losses (Q_C and Q_V. This point is known as the "heat deficit temperature" (HDT), because, at any point below it, the barn is in a heat deficit condition. In other words, the animals produce *less* heat than the barn loses. Unless supplementary (Q_S) or recovered (Q_r) heat is

supplied the air temperature in the barn will drop — as discussed in section 2.3.

Switching off some of the ventilation fans (i.e. reducing the ventilation rate to a level less than the required winter minimum rate) whenever the outdoor temperature is below the HDT is usually not a desirable solution, as previously explained (although a slightly higher barn air temperature will result).

Outdoor air temperatures in Ontario are frequently below the HDT for many barns — for a substantial portion of the winter season. In terms of profits, the consequences of this can be serious, and a barn heat deficit condition should be averted if possible, and in the most economical way. This is where heat recovery comes into play: it is an alternative to supplementary heat.

3.4 NVENT: Supplying the numbers

Calculating heat losses, animal heat production, HDT's and heat recovery values is somewhat complex — and time-consuming. Fortunately, a computer program, called NVENT, has been developed by the Ontario Ministry of Agriculture and Food to do all these calculations for you. In fact, it goes one step further and determines how heat recovery may benefit your barn. The program is available from your agricultural engineer.

To run NVENT, the engineer needs to have certain information about your barn, including the following:

- Livestock type
- Inside temperature
- Number and weight of animals
- Building dimensions
- Amount and type of insulation in walls, ceilings, and foundation
- Type and size of air inlet
- Approximate minimum winter ventilation rate (number and type of fans used during winter)

Referring back to Figure 2-2, the NVENT program will supply values of Q_A, Q_S Q_C and Q_V for your barn. In addition it will also tell you what the HDT is. Knowing these, you can now determine how heat recovery fits into your plans.

4. WHEN SHOULD I CONSIDER USING HEAT RECOVERY?

4.1 Consideration criteria — do I need it at all?

To begin with, heat recovery only makes sense for your barn if it is in a negative heat balance (i.e. if the outdoor air temperature is at or below the HDT) for a high percentage of the winter. This can be easily determined by taking the HDT from the NVENT program and referring to Table 4-1 which shows the number of hours that the outside air temperature is at or below various given temperatures.

As an illustration, suppose a hog finishing barn located in the vicinity of Windsor has a calculated HDT of −19°C (−2.2°F). As shown in Table 4-1, this condition persists for fewer than 32 hours, and it would obviously be inadvisable to spend say 3,000 dollars on heat recovery when a 200 dollar electric heater would provide the required heat for this small amount of time.

Conversely, an egg-layer barn situated in the Thunder Bay region, and having an HDT of −5°C would conceivably benefit from heat recovery for at least 2,334 hours as shown in the table.

4.2 Evaluation criteria — physical performance

Heat exchangers vary both in terms of the **efficiency** with which they capture heat from the outgoing air, as well as the **total amount** of heat they capture. A number of procedures are used to evaluate efficiency but in most cases this is often more of an academic consideration.

For example, a very efficient recovery system will not only recover sensible heat from the outgoing air, but will also condense some of the water vapor. Going one step further, as more heat is extracted from the exhaust air stream, the liquid water condensate will then freeze as additional heat is pulled out, thus causing an ice blockage in the air passage. In this state ventilation and heat recovery are both suspended. Therefore, exchanger efficiency is often deliberately lessened in the design stage to control icing.

TABLE 4-1: TEMPERATURE DURATION DATA FOR SELECTED ONTARIO LOCALITIES.
Number of hours (per annum) at or below given temperature (degrees Celsius)

Location	+5	+2.5	0	−2.5	−5	−7.5	−10	−12.5	−15	−17.5	−20 or lower
Hamilton	3608	3008	2246	1599	1145	791	486	264	125	54	22
Kingston	3757	3021	2448	1864	1390	1061	748	513	311	185	88
Mt. Forest	4199	3651	2966	2259	1692	1251	849	532	300	153	64
Ottawa	4035	3594	2868	2301	1831	1431	1073	759	528	325	181
Simcoe	3698	3141	2356	1603	1159	759	483	258	147	54	15
Sudbury	4545	4060	3426	2758	2295	1845	1486	1113	820	560	346
Thunder Bay	4717	4184	3480	2817	2334	1943	1591	1258	972	709	491
Toronto	3837	3240	2411	1699	1243	861	551	328	194	90	37
Waterloo-Wellington Airport	4157	3221	2465	1786	1288	870	569	327	179	89	33
Windsor	3447	2898	2146	1325	912	561	342	170	66	32	11

Information supplied by Environment Canada, Atmospheric Environment Service.

One useful term developed by the Prairie Agricultural Machinery Institute (PAMI) for expressing heat recovery performance is the heat recovery ratio (HRR). It is equivalent to the amount of energy recovered divided by the total amount of energy being removed (ejected) by the heat exchange device. In mathematical terms it is simply:

$$HRR = \frac{\text{Energy recovered by exchanger}}{\text{Energy available to exchanger}}$$

In the instance of the exchanger shown in Figure 1-2 an HRR of approximately 40% was measured, as was also the case for the unit shown in Figure 1-5 when the outside and barn air temperatures were about −15°C (+5°F) and 18°C (65°F) respectively.

The HRR is not, by any means, a perfect or complete indicator of performance and should be used with care. Nevertheless, it is a convenient way of **comparing** various exchangers, and most reliable vendors of such equipment should be able to readily supply this information.

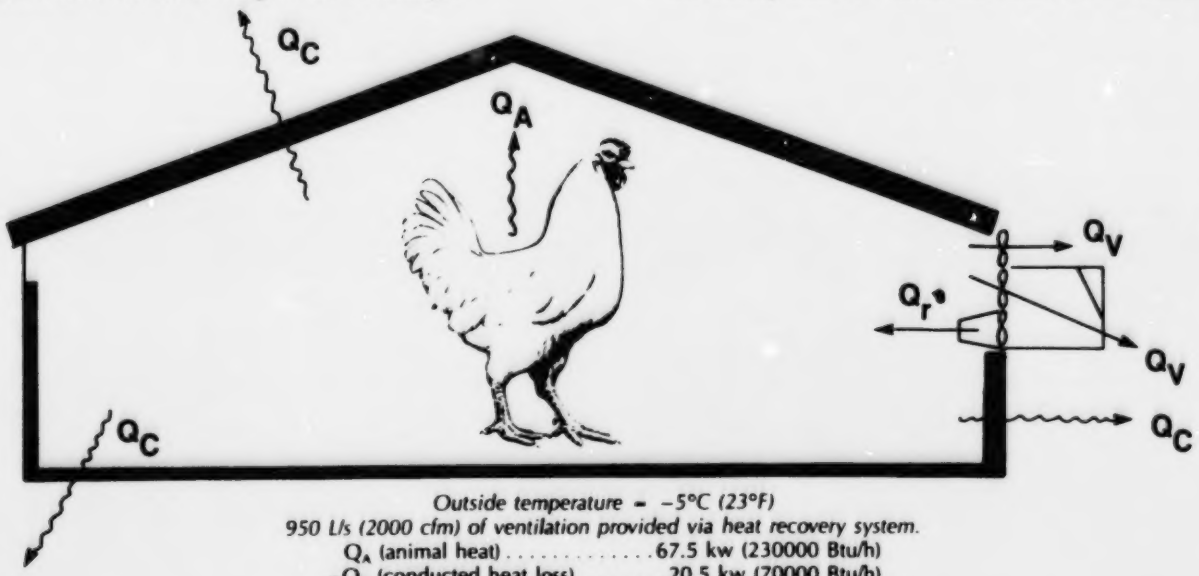
Note that you must know the conditions under which the HRR is measured because it *decreases* as the temperature difference between barn and outside air *increases*. In addition the humidity or "wetness" of the exhaust air also influences the HRR. Any heat exchanger will have a *range* of HRR values, corresponding to different temperature and humidity conditions. Beware of inaccurate claims, and do *not* rely on the HRR as the sole criteria for evaluating a heat exchanger.

The HRR achieved under actual operating conditions, i.e. in the barn, will be less than most published laboratory results due to dust fouling — which diminishes both the rate and efficiency of the heat transfer.

Perhaps a more important number to use in evaluating a heat exchanger for your operation is the actual amount of heat recovered (Q_r), expressed in kilowatts (kW), kilojoules per hour (kJ/h) or sometimes British thermal units (BTU's) per hour. This number is of great significance because the heat being exhausted from the barn must always be exhausted, irrespective of whether it goes through a simple exhaust fan or across a heat recovery device. In otherwords, what really matters is the amount of heat you get **back** from the exhaust air.

Although the single-plate exchanger in Figure 1-2 and the multi-plate exchanger in Figure 1-5 demonstrated comparable HRR's, their heat recovery rates were found to be unequal, the former returning about 2 kW of heat to the barn vs 3.3 kW for the latter. This is probably because, in this instance the multiplate exchanger condensed more water vapor and hence recovered a correspondingly greater proportion of the latent heat contained in the exhaust airstream.

Figure 4-1 depicts the effect of heat recovery in the barn originally shown in Figure 2-2. In this case, half of the minimum winter ventilation rate is being provided by a heat recovery system having an HRR of about 55%. Consequently an additional 16 kW (55,000 BTU/h) of heat is now being returned (recovered) and put to good use



Outside temperature = −5°C (23°F)
950 L/s (2000 cfm) of ventilation provided via heat recovery system.

Q_A (animal heat)	67.5 kw (230000 Btu/h)
$-Q_C$ (conducted heat loss)	20.5 kw (70000 Btu/h)
$-Q_V$ (ventilation heat loss)	63.0 kw (215000 Btu/h)
$+Q_r$ (recovered heat)	16.0 kw (55000 Btu/h)
Heat Balance	0

∴ No heat deficit, barn air temperature held at 22°C (72°F)
Result: optimum feed conversion, maximum profits. (adapted from ref. 6)

Figure 4-1: *Energy budget (with heat recovery)*

in the barn. This 16 kW of heat has a tangible economic value whenever the barn is in a heat deficit condition, i.e. whenever the outdoor air temperature falls below the barn HDT. Without heat recovery the HDT of this barn is actually 2°C (36°F) according to the NVENT program, versus −4°C (25°F) with recovery.

If you have a reasonable estimate of Q_r for a typical set of barn and outdoor conditions, you will be able to calculate an approximate value for the total (seasonal) amount of useful heat a given device will recover. It is then comparatively easy to calculate the dollar value of this recovered energy, whether the savings are in the form of either supplementary heat or improved feed conversion.

4.3 Acceptance criteria — economic performance

In making a final decision regarding the economic advisability of purchasing or building a heat exchanger, it must be remembered that the only unique thing an exchanger does is recover heat. Although claims are sometimes made concerning enhanced ventilation or improved feed conversion, these are indirect benefits that can be obtained without necessarily resorting to heat recovery. Heat exchangers ventilate, but so do simple exhaust fans; heat exchangers temper (warm) incoming air, but a make-up air preheater — either fossil-fuel fired or electrically powered — will do likewise. In some cases, heat recovery has resulted in improved animal health, but this usually was because the facilities had previously been underventilated — as discussed in section 3.1. In short, the benefit derived from heat exchangers is the heat they supply, which displaces supplementary heat that might otherwise be provided.

The most straightforward approach to evaluating a heat recovery system is to compare its cost with that of the fans and heaters needed to provide an equivalent amount of ventilation and heat, and then credit the recovery unit with the dollar value of the energy (or feed) it saves.

One commonly employed technique used to evaluate economic feasibility is payback, which is defined as follows:

$$\text{Payback (years)} = \frac{\text{Purchase price of equipment}}{\text{Annual dollar savings achieved with that equipment}}$$

Thus, if a 2,000 dollar item of equipment saved a farmer 400 dollars annually, its payback would be 2,000/400 = 5 years. Paybacks of less than three years are usually considered attractive, whereas a payback period extending beyond ten years should be approached with some caution, and not accepted in most cases.

Although simple in its approach, the payback concept provides a preliminary indication of whether an investment is worthwhile. More detailed financial analysis is encouraged whenever the grower feels it is warranted.

The following example illustrates how an economic evaluation of heat recovery might be made.

Example problem. A farm operator is considering heat recovery for a weaner room on his Ottawa area farm. The following information applies:

No. of animals . . . . . 180 - 25 kg (55 lb.) weaners
Room temperature . . . . . 25°C (78°F)
Minimum ventilation rate . . . 342 L/s (720 c.f.m.), from Table 3-1
Exchanger system exhaust flow . . . . . 190 L/s (400 c.f.m.)
Exchanger recovered heat . . . . . 3.1 kW (10,600 BTU/h) @ −6°C (20°F) outdoor temperature
Exchanger system cost (including installation) $2,100
Present heat source . . . . . electricity
Energy cost . . . . . $0.057 per kilowatt hour (kWh)

The required information is loaded into NVENT, which calculates the room HDT as being 7°C (45°F). What will the payback on his investment be?

Solution: Referring to Table 4-1, it can be seen that the outdoor air temperature in the Ottawa area is at or below +5°C for 4,035 hours, so the recovery system will provide a usable benefit for *at least* this amount of time. The energy saved (i.e. recovered) by the system is then 3.1 kW x 4,035 hours = 12,508 kWh. Its value is then 12,508 x $0.057 = $713. The payback is $2,100/$713 per year = 3 years.

Note also that the recovery system saves the cost of a 190 L/s fan, a 3 kW heater as well as any additional wiring needed to supply 3 kW of electric heat.

If the heat in the previous example was being provided by propane, costing $0.25/L and being combusted at 75% efficiency, then by referring to Appendix 2 it can be seen that 12,508 kWh of electric heat is equivalent to (12,508 kWh x 3.6 MJ/kWh) / (25.5 MJ/L x 0.75 efficiency) = 2,354 litres of propane, having a value of 2,354 x $0.25 = $588. In this case, the payback is $2,100/588 = 3.6 years.

4.4 Additional criteria — maintenance and air distribution

For the most part, maintenance requirements of heat exchangers may be summed up in two words: dust and ice. With regard to the latter, enough experience has now been acquired in this science to permit virtually any properly designed recovery device to operate in a livestock facility without icing up. Freezing of the water that is condensed (from the outgoing exhaust air) is simply an indication that too much heat is being removed from the exhaust. Be wary of equipment that lacks a guarantee concerning ice-free operation.

Freezing is averted in numerous ways, including: deliberately designing for lower heat transfer efficiency

by, for example, employing wide air passageways or by reducing the total amount of heat transfer surface area; using unbalanced air flows, i.e. by exhausting more air than is being supplied, thus having a "surplus" of heat within the exchanger and; incorporating a defrost cycle in the unit, actuated either by a timer and/or an outdoor thermostat.

This last technique is quite effective and usually involves reversing the supply air fan for a comparatively short period of time during which both it and the exhaust fan are exhausting, i.e. pushing warm air into the exchange device. This will melt any ice deposits that may have formed on the exhaust side of the heat transfer element. The supply fan subsequently resumes its forward (air supply) mode of operation.

Fouling of exchangers due to dust accumulation can be more difficult to contend with. In some types of livestock facilities, such as dairy calf rooms, dust levels in the exhaust air are low enough that significant dust accumulation levels seldom, if ever, occur. A thorough semi-annual rinsing with warm water is typically all that is required. Conversely, the potential for extreme fouling — and even total blockage — by dust exists in many turkey, chicken broiler, egg-layer and hog barns, simply because of higher dust levels within the barn atmosphere.

Fouling by dust is also related to the type of exchanger used. For example, large single-surface exchangers reportedly resist fouling, as would be expected. However, whenever small (i.e. between 0.3 and 2.5 cm or 1/8 inch to 1 inch) air passages are found the possibility of fouling is much greater. In these cases, provision for cleaning *must* be made.

Past experience with filters, similar to those found on residential forced-air furnaces, has been unsatisfactory. Although larger pieces of debris are intercepted by such filters, smaller particles are invariably transmitted and thereafter adhere to the heat transfer surfaces, accumulating until complete blockage occurs. In one instance, a filter-equipped exchanger became extremely fouled within eight weeks of operation, even though the filter element had been manually cleaned on an almost daily basis. Resorting to a very fine mesh filter is not recommended, since this will cause reduced air flows.

Cleaning of exchangers having narrow air passageways is usually accomplished by manual rinsing with water on an as-required basis. Using high-pressure washers may damage some types of exchangers, and you should consult with the supplier before attempting this.

If water-washing is used, the exchanger should be designed so that the wash water and debris may drain out of the exchanger, preferably exterior to the barn. A thorough washing of the heat transfer surface is recommended. In some cases warm water containing detergent has enhanced cleaning efficacy.

Some types of exchangers, notably plate-type plastic matrix designs experience fouling close to the discharge end of the exhaust passages. This may occur even if the entrance to each passage has been thoroughly spray-washed. For this reason a specialized hand-held wash wand, such as that shown in Figure 4-2, is often used to clean the deeper portions of an exchanger core. The wand, constructed of copper tubing contains numerous small holes at the extreme end through which water is directed to assist in dislodging debris. It can be connected to a water hose and operates at standard domestic water supply pressure.

High operating efficiency can be sustained by adhering to a regular cleaning program, incorporating spray-washing and periodic deep-core cleaning with the wash wand.

Figure 4-2: *Hand-held wash wand used for cleaning interior portion of plate type heat exchanger core. (ref. 7)*

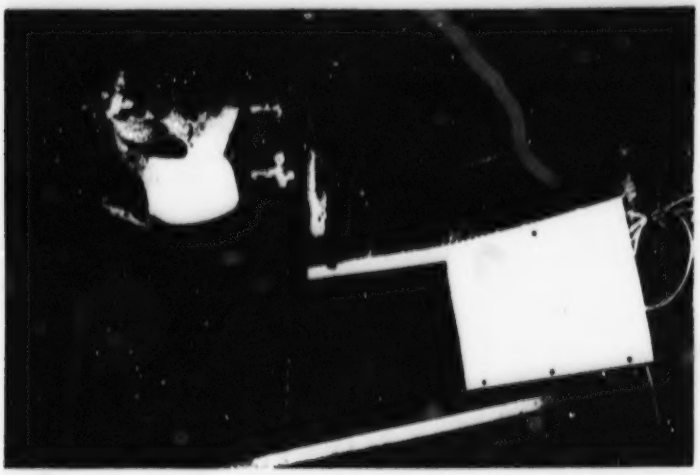


Figure 4-3: *View of plate type heat exchanger interior (hinged fan door open) showing eight spray nozzles used for automatic cleaning. (ref. 7)*

Some producers dislike the inconvenience and backspray associated with manual cleaning. An alternative approach is automatic washing which has been demonstrated to be effective and reliable in maintaining exchanger cleanliness. At present there are no known exchangers possessing factory-equipped washing systems. Consequently, a retrofit approach is, in all likelihood, the only way to acquire an automatic cleaning system.

Figure 4-3 shows a plate type exchanger with four pairs of nozzles fitted to the inside of the hinged fan door. The flow of water through the nozzles may be effected by a simple solenoid valve operated by a time-clock, as seen in Figure 4-4. In this case the nozzles, having a combined flow capacity of 3.6 L/min (1 US gpm), were operated for 2 to 4 minutes each hour, with good results as shown in Figure 4-5 which illustrates the condition of the exchanger core after 21 days of exchanger operation. The exchanger shown in Figure 4-6 was not equipped with a cleaning system and was operated simultaneously and positioned directly beside the exchanger in Figure 4-5.

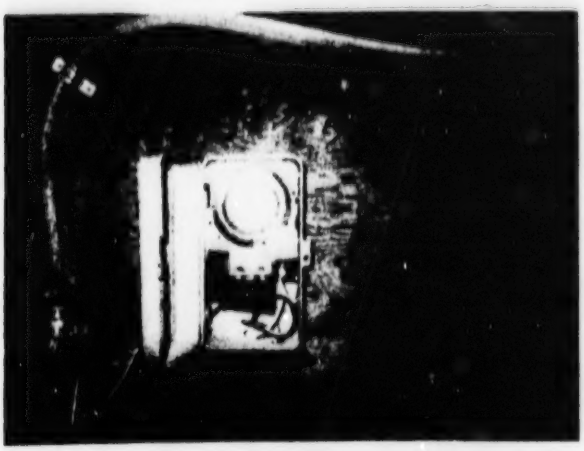


Figure 4-4: *Time clock and solenoid valve used to control automatic cleaning cycle. (ref. 7)*

Subsequent testing for four crops of broiler chickens confirmed that the automatic system was feasible, although slightly longer cleaning durations (i.e. 8 minutes per hour) may be advisable. The cost of such a system was approximately 350 dollars.

Some heat recovery systems can both exhaust and supply air at more than a single location, thus providing effective air distribution and circulation. Single-point exchangers lack this capability and in some instances provision for distributing the fresh air evenly throughout the barn must be made. This is typically done by ducting the air, sometimes to either a preheat room or an air bag. Alternately, certain exchangers "jet" the fresh air into the room at a comparatively high velocity (e.g. 10 m/s or 2,000 ft/minute). This minimizes drafts since the high velocity incoming air stream "entrains" or mixes with warmer room air before losing its momentum and dropping down to the level of the livestock.

Figure 4-5: *Core of automatically-cleaned heat exchanger after 21 days of operation in broiler chicken barn.*

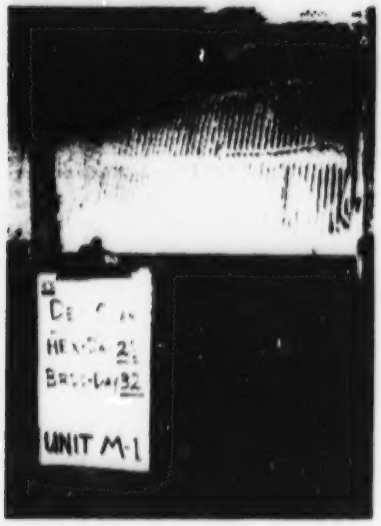


Figure 4-6: *Core of exchanger identical to unit pictured in Figure 4-5, showing comparative appearance after 21 days of operation, but without automatic cleaning.*

5. WHO SHOULD BUILD MY HEAT EXCHANGER?

The answer to this question depends on you. If you are reasonably handy, own the correct tools and workspace, are sufficiently patient and are willing to devote the necessary time and attention needed to construct and start up your own exchanger, then you may be able to achieve some financial savings by taking this approach.

However, you should recognize that some specialized machining and assembly operations are not easily carried out on the farm, so the choice of exchangers suitable for on-farm fabrication is limited. Availability of plans for self-built exchangers may likewise be limited.

To begin answering this question, obtain prices and performance specifications for factory-manufactured exchangers. Then consult with your agricultural engineer to determine which plans for self-built exchangers are available, and how the performance of such a unit will compare with a manufactured one. (Of course you will first have to verify that heat recovery is worthy of consideration — as outlined in section 4.1). Then compile the cost of materials for a self-built exchanger, estimate the labour that will be needed to assemble it, and put a value on your own time, even if it's only a nominal one. Compare the estimated cost of your self-built unit to a commercially-manufactured exchanger.

Then decide if you are going to build it, or buy it.

6. WHERE CAN I OBTAIN MORE INFORMATION (OR HELP) CONCERNING HEAT RECOVERY?

This booklet provides the necessary information for making a basic technical and economic appraisal of heat recovery for your livestock operation. Making this appraisal is the most important step in the pathway to heat recovery.

By first answering the six questions in the booklet; what, why, how, when, who and where, you will have prepared yourself to answer the crucial seventh question, which is: "Will heat recovery improve the profitability of my enterprise?"

The two principal sources of additional information on this subject are equipment dealers who sell heat recovery devices and agricultural engineers knowledgeable in this area, working either for the Ontario Ministry of Agriculture and Food or in private practice.

Obviously, most suppliers and professional advisors have their own preferences. The information contained in this booklet will assist you in determining whether or not you concur with their preferences.

You may know of a neighbour who has already installed heat recovery equipment in his/her barn. His/her experience will likely assist you in evaluating heat recovery for your operation.

Finally, the references and additional information sources listed at the back of this booklet provide more insight into the subject. You might find the pursuit somewhat exhausting, but worthwhile as much information is now available.

APPENDIX 1: USEFUL CONVERSION (REF. 3)

Metric Terms

km — kilometre
m — metre
mm — millimetre
m^2 — square metre
cm^2 — square centimetre
L/s — litre per second
L — litre
m/s — metre per second
kg — kilogram
g — gram
W — watt
c — centigrade
RSI — Resistance System International
Pa — Pascal

Conversion Figures

Imperial	Metric	Imperial	Metric
1 mile	1.6 km	1 cubic foot (gases)	28.3 L
1 yard	.914 m	1 foot per second (Note per second)	.304 m/s
1 foot	.305 m	1 pound	.45 kg
1 inch	25.4 mm	1 ounce	28.3 g
1 square yard	.836 m^2	1 BTU/h	.293 W
1 square foot	.092 m^2	(C° × 9/5) + 32°F	C°
1 square inch	6.45 cm^2	1 R	.176 RSI
1 cubic foot per minute (cfm)	.471 L/s	.125 inches (Static Pressure)	31 Pa
1 gallon (Can)	4.5 L		
1 quart (Can)	1.13 L		
1 pint (Can)	.568 L		

APPENDIX 2: ENERGY CONTENT OF SOME ENERGY SOURCES

Energy Source	Energy Content — Metric Measure	Energy Content — Imperial Measure
1. Electricity	3.6 MJ/kWh	3,412 BTU/kWh
2. Gaseous Fuels		
Natural Gas	37.3 MJ/m^3	1,000 BTU/cubic foot
Propane	25.5 MJ/L	109,800 BTU/gallon
3. Liquid Fuels		
Diesel Fuel No. 2 Heating oil	38.6 MJ/L	166,500 BTU/gallon
Gasoline	34.8 MJ/L	150,000 BTU/gallon
4. Solid Fuels		
Wood — air dried ≃15% m.c.	16.3 MJ/kg	7,000 BTU/lb.
— white ash, hard maple+	≃25.3 GJ/cord*	≃24 × 10^6 BTU/cord*
Coal — bituminous	24.4-33.7 MJ/kg	10,5000-14,500 BTU/lb.

*Cord = pile of wood (1.22 m × 1.22 m × 2.44 m, i.e. 3.63 cubic metres of piled wood) (4' × 4' × 8', i.e. 128 cubic feet of piled wood)
+Air dried, seasoned wood.
m.c. = moisture content
Note: For heating purposes, electricity has an efficiency of 100%. Combustion efficiencies: (for gaseous, liquid and solid fuels) may range between 50% and 95%, i.e. 50% to 95% of the heat content of the fuel is actually delivered to the barn as useful heat; the remainder is lost in the exhaust.

APPENDIX 3: EXAMPLE CALCULATION

Example calculation: Egg-layers (caged)

Parameters

The egg-laying operation referred to in figures 2-2 and 4-1 and section 3.2 has the following parameters:

No. of birds . . . . . 10,000 @ 1.8 kg (4 lb.)
Building location . . . . . Windsor
Building dimensions . . . . . 76.2 cm x 9.8 m x 2.4 m (250 ft. x 32 ft. x 8 ft.)
Desired indoor air temperature . . . . . 22°C (72°F)
Insulation — ceiling . . . . . RSI 5.3 (R-30)
— walls . . . . . RSI 5.3 (R-20)
— foundation . . . . . none
Feed consumption (Nov. to Mar.) . . . . . 162 tonnes
Present feed conversion (Nov. to Mar. inclusive) . . . . . 1.68 kg (3.7 lb.)/dozen eggs
Production 77%

Ventilation rate

At present, the operator sometimes underventilates the barn in an attempt to maintain higher air temperatures. He/she feels that 0.5 L/s per hen, i.e. 5,000 L/s (10,500 c.f.m.) as suggested in Table 3-1 will provide far too much ventilation, and so decides instead to specify a minimum ventilation rate sufficient to provide 4 air changes per hour, calculated in Section 3-2 to be about 1,990 L/s (4,220 c.f.m.).

HDT and heat recovery

The NVENT program is then used, with results as shown in the sample printout. The HDT, at 1,992 L/s (4,220 c.f.m.) is computed to be about 2°C (36°F).

Referring to table 4-1, it is determined that the outdoor temperature in the Windsor area will on average be at or below this HDT for some 2,700 hours, or roughly 3/4 of the total November to March period of 3,500 hours.

Again referring to NVENT, it can be seen that by adding 950 L/s (2,000 c.f.m.) of heat recovery at 55% HRR, in the order of 16 kW (55,000 BTU/h) of heat can be returned to the barn when the outside air temperature is −5°C (23°F). (More kilowatts are recovered as the outdoor temperature drops even lower — as shown on the sample NVENT printout).

The net effect is that with heat recovery, a 6 C° (10 C°) *higher* inside air temperature can be sustained, if required. (Note that heat recovery is not needed above the HDT).

Estimated savings

Referring to Figure 2-1, it can be seen that feed consumption (per dozen eggs) *increases* by about 0.028 kg for each degree Celsius below 22°C. Consequently, a 6°C temperature increase will save 0.028 x 6 = 0.168 kg of feed per dozen eggs. However, since the barn does not operate below the HDT for the entire November to April period, the farmer estimates his/her savings will be closer to 2/3 this amount (i.e. the barn will balance at 22°C without requiring heat recovery for the remaining 1/3 of the winter).

His/her preliminary estimate of feed saving is then 2/3 x 0.168 = 0.112 kg/dozen. The total saving is estimated to be 0.112 kg/doz. x 642 doz. x 150 days = 10,785 kg or about 10.8 tonnes.

To be extremely conservative, the farmer reduces this by 25% to allow for uncertainty, thus arriving at a final estimated feed savings of 8 tonnes, i.e. a 5% reduction, yielding a corresponding improvement in hen performance, from 1.68 kg of feed per dozen eggs down to 1.60.

Note also that he/she is now able to ventilate the barn adequately, in contrast to the previous situation wherein humidity and noxious gas levels were, at times, excessive.

Payback

The farmer is considering employing 8 single-plate exchangers, similar to that pictured in figure 1-2. Each unit will exhaust approximately 120 L/s (250 c.f.m.) and recover 2 kW (6,800 BTU/h) of heat. The costs are estimated as follows:

Materials	$2,400
Fans	2,900
Labour (160 h)	1,600
Allowance for miscellaneous costs	1,000
TOTAL	$7,900

The present feed price is about $190 per tonne. The payback is then $\frac{7,900}{1,520}$ = 5.2 years.

Since he/she anticipates providing the labour personally, the corresponding cost component is, in effect, a wage, which enhances the attractiveness of the project.

Other notes

It should be added that the savings for an identical barn situated in Mt. Forest or Thunder Bay, for example, would be higher since a greater percentage of the winter season is spent at or below the HDT. (See table 4-1).

Finally, it is conceivable that the cost of commercially manufactured equipment might, in fact, be less than that of the self-built alternative. Thus for example, the grower may discover that 2 suitably sized unit type exchangers could be purchased for a total cost of $6,000 including automatic washing systems and installation. This would improve the payback to $\frac{6,000}{1,520}$ = 3.9 years.

It is important to compare prices and performance characteristics before making a final decision. In short, answer the six questions.

Addenda:

APPENDIX 3: SAMPLE NVENT PRINTOUT.

```
------------------------------------------------------------
 CAUTION:   THIS PROGRAM REQUIRES INTERPRETATION BY AN ENGINEER
            OR OTHER PERSON WITH A THOROUGH KNOWLEDGE OF THE
            PRINCIPLES OF VENTILATION AND HEAT BALANCE.
------------------------------------------------------------
      * * * * * * * * * * * * * * * * * * * * * * * * * *
      *                                                 *
      *  Authorized for release by:                     *
      *                                                 *
      *  _____________________________                  *
      *  Agricultural Engineer                          *
      *                                                 *
      *  Date:                                          *
      *        ----------------------                   *
      * * * * * * * * * * * * * * * * * * * * * * * * * *

      * * * * * * * * * * * * * * * * * * * * * * * * * *
      *                                                 *
      *    ONTARIO MINISTRY OF AGRICULTURE & FOOD       *
      *      AGRICULTURAL ENGINEERING SERVICES          *
      *                                                 *
      * * * * * * * * * * * * * * * * * * * * * * * * * *

      ------------------------------------------------
                   VENTILATION  PROGRAM
                      VERSION  1.2
                     DATE  86-11-13
      ------------------------------------------------

 FRESH AIR FARMS     88  3  1

NVENT SAMPLE: ENERGY TECHNOLOGIES
---------------------------------

 4) Livestock type ..............................  POULTRY
 5) Inside temp. and relative humidity ..........   22.0C @  75%
 6) Outside design temp. and relative humidities
    below and above freezing ....................  -15.0C @  80%,  80%
 7) Number and weight of:  A) Small Animals .....  10000 @ 1.8 kg
 8)                        B) Medium Animals ....     0 @  0.0 kg
 9)                        C) Large Animals .....     0 @  0.0 kg
10) Building length, width, and height ..........  76.2 m  x  9.8 m  x 2.4 m
11) RSI value of walls ..........................  RSI  3.5
12) RSI value of ceiling ........................  RSI  5.3
13) Doors: Total area, RSI value ................   8.3 sq.m , RSI  0.9
14) Windows: Total area, glazing ................   0.3 sq.m , 2 pane
15) Thickness of foundation insulation ..........  0 mm
16) RSI value of floor ..........................  RSI  1.8
17) Temperature below floor .....................  10.0 C
20) Type of air inlet ...........................  SIDE AIR INLET
21) Total length of inlet .......................  76.0 m
22) Inlet velocity desired ......................    4.1 m/s
```

HEAT BALANCE AT -15 C AND RECOMMENDED VENTILATION RATE OF 1331 L/s

COMPONENT		HEAT LOSS (Watts)
EXTERIOR WALLS		4273
INTERIOR WALLS		0
CEILING		5213
DOORS		341
WINDOWS		35
FOUNDATION		9037
FLOOR	+	4978
TOTAL BUILDING HEAT LOSS		23878
VENTILATION HEAT LOSS	+	57640
TOTAL HEAT LOSS		81518
HEAT GAIN FROM ANIMALS		68117
HEAT BALANCE (HEAT GAIN - HEAT LOSS)		-13401

THE TOTAL ANIMAL MOISTURE PRODUCTION AT THE
INSIDE DESIGN TEMPERATURE IS APPROXIMATELY 65 kg/hr

VENTILATION REQUIREMENTS

OUTSIDE TEMP. (C)	FAN CAPACITY FOR MOISTURE CONTROL (L/s)	FAN CAPACITY FOR TEMP. CONTROL (L/s)	RECOMMENDED FAN CAPACITY (L/s)	AIR INLET SETTING (mm)	SUPPLEMENTAL HEAT REQUIRED (Watts)
-30	1262	597	1262 *	4.05	40808
-25	1275	707	1275 *	4.09	31453
-20	1296	844	1296 *	4.16	22301
-15	1331	1021	1331 *	4.27	13401
-10	1387	1258	1387 *	4.45	4795
-5	1480	1593	1593 *	5.11	0
0	1645	2109	2109	6.77	0
5	1906	2741	2741	8.80	0
10	2441	4082	4082	13.10	0
15	3932	7354	7354	23.60	0
20	17876	25725	25725	82.56	0
25	17176	26622	26622	85.44	0
30	14007	27697	27697	88.89	0
35	5673	29017	29017	93.12	0

* FAN CAPACITY IS LESS THAN 1991 L/s (4 AIR CHANGES PER HOUR)

A HEAT DEFICIENCY WILL OCCUR AT TEMPERATURES BELOW -7 C

VENTILATION REQUIREMENTS: SPECIFIED MINIMUM VENTILATION RATE= 1992 L/s
==========

OUTSIDE TEMP. (C)	FAN CAPACITY FOR MOISTURE CONTROL (L/s)	FAN CAPACITY FOR TEMP. CONTROL (L/s)	RECOMMENDED FAN CAPACITY (L/s)	AIR INLET SETTING (mm)	SUPPLEMENTAL HEAT REQUIRED (Watts)
-30	1262	529	1992	6.39	89716
-25	1275	632	1992	6.39	75287
-20	1296	760	1992	6.39	60793
-15	1331	925	1992	6.39	46200
-10	1387	1146	1992	6.39	31461
-5	1480	1459	1992	6.39	16512
0	1645	1941	1992	6.39	1262
5	1906	2532	2532	8.13	0
10	2441	3785	3785	12.15	0
15	3932	6842	6842	21.96	0
20	17876	24081	24081	77.28	0
25	17176	24921	24921	79.98	0
30	14007	25927	25927	83.20	0
35	5673	27163	27163	87.17	0

A HEAT DEFICIENCY WILL OCCUR AT TEMPERATURES BELOW 2 C

VENTILATION REQUIREMENTS: SPECIFIED MINIMUM VENTILATION RATE= 1992 L/s
========== HEAT EXCHANGER CAPACITY= 950 L/s
HEAT RECOVERY RATIO= 55 %

OUTSIDE TEMP. (C)	FAN CAPACITY FOR MOISTURE CONTROL (L/s)	FAN CAPACITY FOR TEMP. CONTROL (L/s)	RECOMMENDED FAN CAPACITY (L/s)	AIR INLET SETTING (mm)	SUPPLEMENTAL HEAT REQUIRED (Watts)	SUPPLEMENTAL HEAT REQUIRED WITH HEAT EXCHANGER (Watts)
-30	1262	529	1992	6.39	89716	57678
-25	1275	632	1992	6.39	75287	46364
-20	1296	760	1992	6.39	60793	35001
-15	1331	925	1992	6.39	46200	23566
-10	1387	1146	1992	6.39	31461	12023
-5	1480	1459	1992	6.39	16512	326
0	1645	1941	1992	6.39	1262	0
5	1906	2532	2532	8.13	0	0
10	2441	3785	3785	12.15	0	0
15	3932	6842	6842	21.96	0	0
20	17876	24031	24081	77.28	0	0
25	17176	24921	24921	79.98	0	0
30	14007	25927	25927	83.20	0	0
35	5673	27163	27163	87.17	0	0

A HEAT DEFICIENCY WILL OCCUR AT TEMPERATURES BELOW -4 C

Errata.
Page 3 Note: Fig 1-4 all dimensions millimeters.

Page 10 Table 2-2 Feed Conversion, kg feed /kg grain should read
Feed Conversion,kg feed/kg gain.

Page 20 Paragraph "HDT and heat recovery"
In the last sentence, 6C° (10C°) should read 6C° (10°F)

NOTES

NOTES

NOTES

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